Use of concept modelling for online input force estimation

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Abstract

Over the years, concept models have found increased use in the design of mechanical systems, such as automotive applications. However, these models can also be exploited as low-cost counterparts of high-fidelity models for use in online applications such as virtual sensors and control. In this work the use of structural models for input force estimation is discussed. Two different approaches are compared to model a twistbeam rear-suspension for estimating the vertical input forces. the first model is an equivalent trailing arm description and the second is a modal reduced model. An experimental setup in which the forces are measured through a Kistler cell is used to estimate the vertical input force from an optical tracking system. The equivalent trailing arm model is shown to be incapable to produce accurate results due to the strong coupling of the vertical motion with the non-vertical loads, whereas the modal reduced model accurately reproduces the vertical forces.

1 Introduction

During the last two decades, automotive OEMs introduced an increasing level of mechatronic content in passenger cars. After saturating the development of passive safety systems (e.g. seat belt, airbags, etc.), and more recently of the Active Safety Systems (ASSs) (e.g. ABS, TCS, ESP, etc.), further advances are expected by the introduction of the Advanced Driver Assistance Systems (ADASs). Complementary to the goals of active safety vehicle control systems, an ADAS uses environment sensors to improve driving comfort and traffic safety by assisting the driver in recognizing potentially dangerous traffic situations [1]. For both ASSs and ADASs, X-in-the-Loop simulations (e.g. Mil as Model-in-the-loop, SiL as Software-in-the-loop, or HiL as Hardware in the loop) are a common practice for designing and testing the control logic implemented in the electronic control units (ECU). To this end, a concept modelling approach was recently proposed to anticipate the mechatronics validation to the concept design phase [2]. This paper deals with the use of concept modelling for only applications. The main goal is to enhance current ADAS and/or ASS applications, by introducing a virtual sensing stage (see fig. 1 right) in the conventional control logic (see fig. 1 left) . This virtual sensing stage is able to provide the controller with an increased range of information regarding the status of quantities that are hardly accessible by conventional sensors. This virtual sensor stage relies on a Kalman filtering approach [3, 4] coupled to a physics based concept model. The coupling of these filters with high-fidelity models recently proved to lead to valuable results [5, 6].

In order to enable realistic computational loads, the use of simple concept models is required. Full finite element models are too computationally expensive to use in estimators in which full time-series are analyzed. However, in order to achieve accurate results, the concept models introduced in this work are based on high-fidelity finite element models. For obtaining accurate results from an estimator a good correspondence between the real system and the model is required, and this is obtained by starting from a finite element

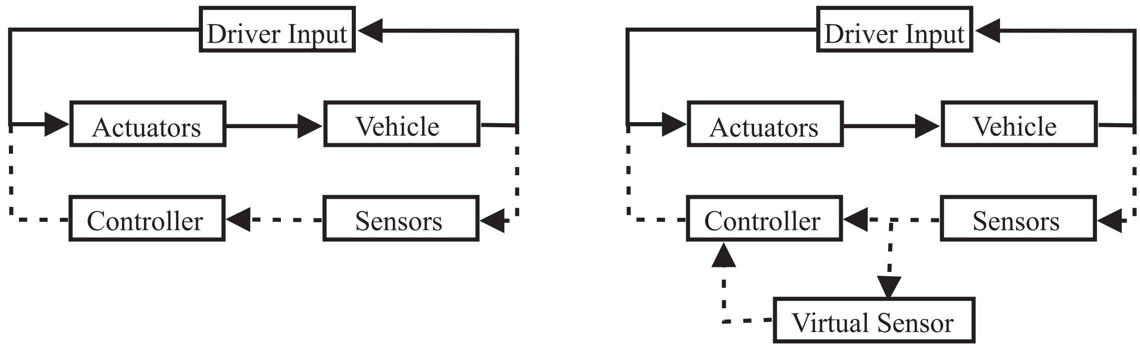

Figure 1: Left: conventional control logic of an active control methodology. Right: Proposed improvements of the control, with the addition of the virtual sensing stage

model which can be updated through classical model updating approaches based on modal analysis. In a second stage a concept model of this system is derived. In this work two different concept models are compared:

- an equivalent one degree-of-freedom model for the unknown force of interest;
- a modal reduced model taking all input forces into account.

In order to obtain the estimates for the unknown input forces, the model has to be used in an estimator. Over the past decades a wide range of estimators have been proposed in literature [4]. In this work the focus is on the use of Kalman filters [3] with augmented states for the unknown inputs. This approach provides accurate results for linear systems, like the structure used in this work, and has a very low computational load. Moreover, this approach is also applicable in the case of dynamic and transient load cases because the filter takes the dynamic behavior of the system into account, which is not the case in classical model inversion for force identification.

The use of the proposed approach is validated experimentally on an industrial case study involving a twist-beam rear suspension (shown in fig. 2), where the goal of the virtual sensor is to estimates the unknown vertical input forces at one of the wheel centers of the suspension. For the validation, the structure is attached to a powerful six-axial shaker available at the research group from the KU Leuven, which allows excitation of both large deformations and small high frequency vibrations. The connection points to the subframe and one knuckle are fixed such that the twisting can easily be excited. A comparison between the measured input force and the virtually sensed one clarifies the potential of the presented approach.

Focus in this work is on estimating the vertical force applied to the twistbeam. This could be valuable information for e.g. an (semi-)active suspension system. However, the proposed techniques can also be used for estimation e.g. lateral forces during cornering.

2 Concept model of twistbeam suspension

The estimator proposed in this work requires a model to infer the unknown applied forces from a given set of measurements. A straightforward approach would be the construction of a single mass-spring-damper system which enables the inference of a single force of interest. However, as will be shown in Sec. 4, combined loading on many mechanical structures requires a multidimensional model to properly capture the behavior of the system. Therefor a concept model based on a finite element model of the structure is proposed in this work. The concept model is constructed by projecting the original model onto its static response modes for the different input points. In the next sections both the original FE model and the concept model are discussed more in detail.

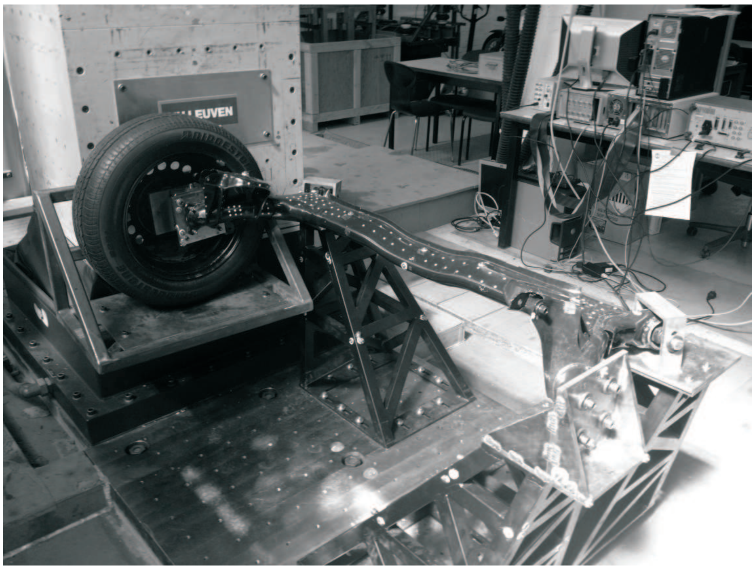

Figure 2: Twistbeam rear suspension

2.1 Finite elements model

The original model from which the concept model is derived, is a linear finite element model of the system under investigation. The construction of this model consists of two main steps:

- **Model construction** In a first phase the basic layout and the type of elements for the model have to be chosen.
- **Model updating** In a second phase the model has to be updated to match the real-life behavior of the physical system.

These two steps are discussed more in detail in the following sections.

2.1.1 Model construction

In the first step of the high-fidelity model construction, a geometry is created from design drawings. This geometry serves as the starting point to perform the finite element meshing. Once this geometry is defined, an automated meshing tool can be applied to mesh the given geometry. An important choice in the creation of the finite element model is the choice of the type of elements. Most general approach is the use of solid elements, but this leads to an unacceptable number of elements for many systems. In automotive applications many if not most components are constructed from sheet metal. In this case a shell mesh is more suitable to generate the model. An important advantage of using shell meshes for these components is the fact that these are also well suited for performing model updating. In the case of a solid mesh, the mesh has to be adapted to enable thickness variations for the different parts (e.g. due to manufacturing tolerances). However, in the case of shell elements, there is a simple dependence between the model matrices and the thickness. For these reason a shell mesh is constructed for the twistbeam system considered in this work, as shown in Fig. 3.

A final important choice in the creation of the finite element model, is the choice on how to connect the different parts of the geometry. The most straightforward approach is the merging of adjacent nodes. However, this approach has two important drawbacks in practice:

- The meshes of different parts of a geometry are often incompatible when automatic meshing tools are used. Creating compatible meshes typically requires considerable user input.

Figure 3: Shell mesh for the twistbeam rear suspension.

- The connection between different parts are in practice often obtained through welding processes. In this case the characteristics of the connections can be highly uncertain and might require considerable updating. However, if node merging is used it is not straightforward to tune the connection to these characteristics.

An alternative approach is the use of multipoint constraints [9]. In this approach the behavior of several nodes on a part are linked to the motion of a center node and then the center nodes of different parts can be connected together. Several different mathematical descriptions of these constraints exist, but when using commercial software often a rigid connection is used between the nodes. This is also the case in this work, and these multipoint are also displayed in Fig. 3. By tuning the number and location of nodes to which the center nodes are connected, the behavior of the connection can easily be adjusted to match the experimentally observed behavior.

The behavior of the component can be described using the equations of motion for this finite element model:

$$M_{fe}\ddot{q}_{fe} + C_{fe}\dot{q}_{fe} + K_{fe}q_{fe} = B_{fe}F_{ext}. \tag{1}$$

In this equation M_{fe}, C_{fe}, K_{fe} and B_{fe} are respectively the mass, damping, stiffness and input matrix for the finite element model. In the given work these matrices are assumed constant. q_{fe} is the displacement vector of the finite element model and is typically very large in size, leading to unacceptably long simulation times for this model, which indicates the necessity of using concept models. Finally F_{ext} are the external forces applied to the component.

2.1.2 Model updating

The second phase in the model generation is the model updating. Good correspondence between the model and the real-system are required in order to get accurate results from an estimator. In this case the model updating procedure takes different set of reference measurements into account:

- free-free modal measurements;
- clamped static deformation measurements.

The free-free modal measurements give a good view on the ratio of stiffness to mass and how the properties are distributed over a given body. However, they do not necessarily lead to accurate static stiffness results. For this last reason also the clamped static deformation is taken into account, because this could otherwise highly impact the estimated quasi-static forces. The free-free modal analysis is performed using two shakers in order to get enough energy in the system and excite the different mode shapes properly. The frequency range considered is up to 120Hz and the response is captured with eighteen 3D accelerometers. The modal tests are performed and processed using LMS Test.lab [10]. For the static tests the twistbeam is mounted on its supports and different masses are applied at the free end to simulate a vertical load. The response of the twistbeam is captured with a Krypton K600 tracking system, which provides accurate position information. Two main sets of parameters are taken into account in the updating process:

- shell thicknesses: due to the bending of the plates the thickness varies over the different parts;
- multipoint constraints: the welding process creates a stronger connection between some points and the nodes which are coupled together are updated.

These parameters are updated using Optimus [12]. Table 1 gives an overview the measured eigenfrequencies, initial model frequencies and updated model eigenfrequencies. This table clearly shows there can be big

nr.	measured [Hz]	nominal model [Hz]	updated model [Hz]
1	27.0	25.8	26.1
2	55.8	76.0	56.7
3	68.7	78.2	68.5
4	92.2	93.0	91.8
5	101.6	101.5	100.9

Table 1: Measured and model eigenfrequencies

differences in the behavior of the structure due to small model uncertainties, . This demonstrates the necessity for model updating. The updated finite element has a behavior which is very close to the physical system and can be exploited as a starting point for estimation.

2.2 Concept model

As discussed in the introduction, two different concept models for estimating the vertical force acting on the twistbeam are compared in this work. These two models are discussed in more detail here.

2.2.1 Trailing arm model

This is the most basic model which can be constructed for the twistbeam rear suspension. This is a one degree-of-freedom model where one side of the twistbeam is represented by a single swing arm with, as shown in Fig. 4. In the model considered here, the vertical displacement of the force application point z_F is considered as the degree-of-freedom, but alternative choices like the rotation angle are possible as well. This model has only two parameters to determine:

- Stiffness k: the stiffness is set by applying a unit load to the finite element model and matching the vertical displacement for the application point on the trailing arm model.
- Mass m: the mass is determined by matching the first eigenfrequency for the given stiffness of the model.

Since this model has only one DOF and one eigenfrequency, the dynamic force estimation will be limited to input frequencies below the second eigenfrequency of the system.

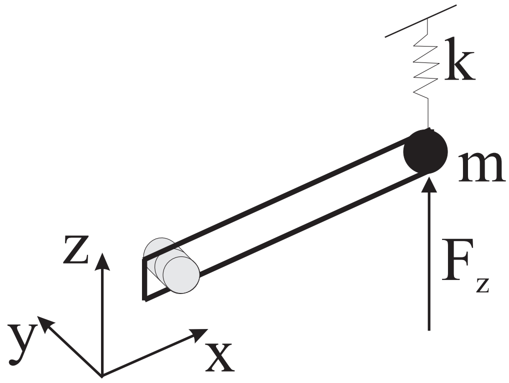

Figure 4: Trailing arm concept model for the twistbeam.

The main difference between the trailing arm model and a regular single mass-spring-damper model, is that this model also allows to represent multiple measurement points on the twistbeam. In order to obtain the measurement equations for point i, the measurement equation becomes:

$$z_i = \frac{x_i - x_0}{x_{ref} - x_0} z_F \,. \tag{2}$$

In this equation the expected vertical motion of a point i scales with the vertical displacement of the force application point and the respective position to the rotation center where the bushing connect the twistbeam to the vehicle frame. The x-coordinates can be obtained from the finite element mesh.

2.2.2 Modal reduced model

The second concept model which is considered in this work is a reduced model by projecting the finite element stiffness matrices onto a set of dominant deformation modes. By projecting on a set of dominant deformation modes, an accurate model can be obtained for a given application at a drastically reduced cost. In literature, a wide range of possible choices for the deformation modes have been discussed [7].

In this work the proposed reduction basis V consists of the first eigenmode of the system V_e, six input modes for the wheel connection V_i and one static deformation mode for the gravity contribution V_g:

$$V = \begin{bmatrix} V_e & V_i & V_g \end{bmatrix} . \tag{3}$$

This basis is used to obtain reduced mass, stiffness, damping and input matrices from the original finite element matrices:

$$M_r = V^T M_{fe} V, \tag{4}$$
$$K_r = V^T K_{fe} V, \tag{5}$$
$$C_r = V^T C_{fe} V, \tag{6}$$
$$B_r = V^T B_{fe}. \tag{7}$$

These matrices are much smaller in size than the original finite element matrices and can be evaluated very efficiently. This makes them perfectly suitable for use in estimation problems [8].

The measurement equations for the displacement for several points i can be evaluated by performing the reverse projection:

$$z_i = V_i q_r \,, \tag{8}$$

whereas q_r is the reduced coordinate vector.

3 Estimator approach

For the estimator, a Kalman filtering approach is adopted [3, 4]. The Kalman filter is a recursively optimal filter for linear systems with white noise disturbance. The main benefit of this filter is the fact that optimality can be achieved at a very low computational cost, making it ideal for online applications. The equations for the discrete time Kalman filter are:

$$\begin{aligned} x_{k+1} &= A_d x_k + B_d F_{known}, && (9)\\ P_{k+1} &= A_d P_k A_d^T + R_x, && (10)\\ K_{kal} &= P_{k+1} H^T \left(H P_{k+1} H^T + R_{meas} \right), && (11)\\ x_{k+1} &= x_{k+1} + K_{kal} \left(y_{k+1} - H x_{k+1} \right), && (12)\\ P_{k+1} &= (I - K_{kal} H) P_{k+1} (I - K_{kal} H)^T + K_{kal} R_{meas} K_{kal}^T. && (13) \end{aligned}$$

This is a predictor corrector scheme where a prediction for the new states x_{k+1} is made first and then corrected by considering the difference with respect to the measurements y_{k+1}. In these equations A_d is the discretized system matrix, P is the estimated state covariance, R_x is the model covariance, H is the measurement matrix, R_{meas} is the measurement covariance and K_{kal} is the Kalman gain. The known or measured input force F_{known} are also taken into account through the discretized input matrix B_d

It is immediately apparent that the Kalman filter if formulated for a time discretized system A_d and B_d. However the equations of motion, as described in the previous section, are described in continuous time, so a time discretization has to be performed. In this work an exponential solver with zero-order hold for the inputs is used to discretize the system from the continuous state equations:

$$\begin{aligned} A_d &= e^{A\Delta t}, && (14)\\ B_d &= A^{-1}(A_d - I) B_x. && (15) \end{aligned}$$

This approach leads to a stable integrator which allows larger timesteps than typical explicit integrators.

In this work an augmented state approach is adopted for estimating the unknown input forces. In this approach the state vector with the position q and velocities $\dot{q}$ is augmented with the unknown input forces F_{un}:

$$x = \begin{bmatrix} q \\ \dot{q} \\ F_{un} \end{bmatrix}. \tag{16}$$

By including these additional states, the forces can be estimated in the same way as the regular degrees-of-freedom. In this case also a model for the forces is required. In this work a random-walk model is adopted which allows maximum freedom in the variation of the forces, at the cost of a larger uncertainty [6]. With this force model the continuous state matrices become:

$$\begin{aligned} A &= \begin{bmatrix} 0 & I & 0 \\ -M^{-1}K & -M^{-1}C & M^{-1}B_{un} \\ 0 & 0 & 0 \end{bmatrix}, && (17)\\ B_x &= \begin{bmatrix} 0 \\ B_{known} \\ 0 \end{bmatrix}. && (18) \end{aligned}$$

With these equations the concurrent estimation of the states and the forces can be performed.

4 Experimental validation

The proposed approach is validated on a twistbeam rear suspension test setup, shown in Fig. 2. In the following sections, first the setup is described more in detail and then the validation results are discussed.

4.1 Setup description

For the experimental validation, the twistbeam structure is fixed at the two bushings and clamped at one of the wheel mounts, as shown in Fig. 2. At the free end a wheel is mounted and between the wheel and the twistbeam a Kistler cell is mounted to record the input forces acting on the twistbeam. The wheel is excited using the Cube, which is a six-axial large motion excitation table. This setup allows the application of large forces over a wide frequency band.

Several motion profiles are applied as excitation to evaluate the proposed approach, and for this work a pulse input profile is analyzed in detail. The profile considered here applies a vertical displacement of $4mm$ and a duration of $2sec$. The forces and torques recorded during this motion are shown in Fig. 5.

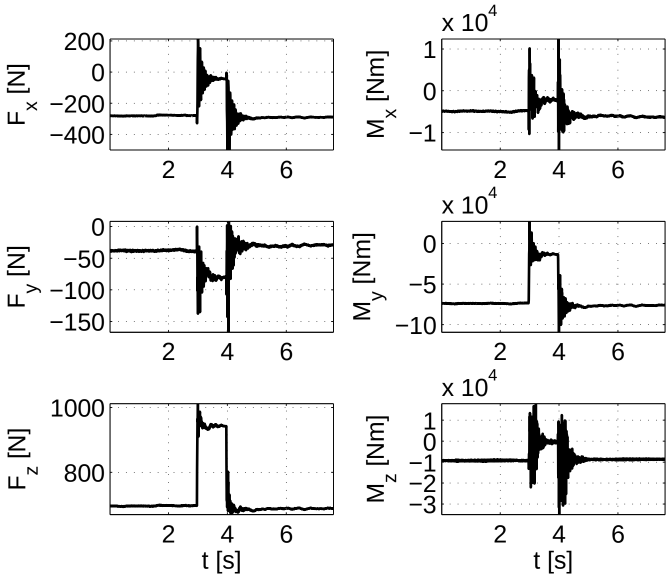

Figure 5: Measured forces for pulse input.

The measurements used for the Kalman filter come from a Krypton K600 system [11]. This system allows tracking of the motion of LED markers on the twistbeam in 3D. In this work however, only the vertical motion is considered. Because the trailing arm only models a part of the twistbeam, only points on this part can be tracked. The modal reduced model on the other hand allows the evaluation of points all over the twistbeam structure. The points used for the measurements are marked in Fig. 6.

Finally the tuning for the Kalman filter has to be performed. In the case of input force estimation in general, the uncertainty on the model is relatively negligible with respect to the uncertainty on the force, such that is suffices to tune this uncertainty. For the given case, the model covariance is set as:

$$R_x = \begin{bmatrix} 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 10^{12} \end{bmatrix}. \tag{19}$$

Because the modal reduced model is also affected by the other input forces besides the force in the vertical direction, these are applied as known external forces. Future research will focus on how these forces can also be estimated in a consistent manner.

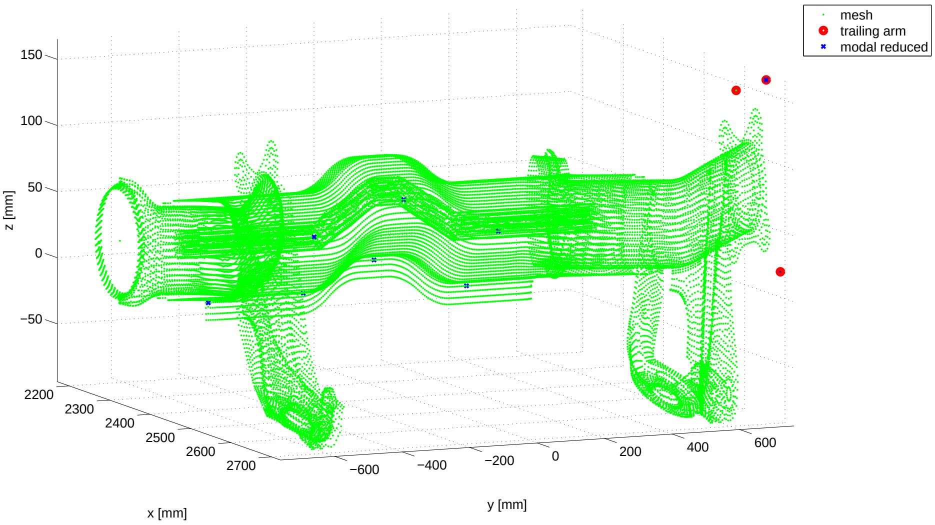

Figure 6: Locations of the measurement points on the twistbeam structure.

4.2 Results

The first aspect to consider for the estimation problem is the tracking of the measurements. Fig. 7 provides an overview of the displacement measurements for the different markers on the twistbeam. A first important aspect is the fact that the trailing arm model can only track the last three measurement points, as mentioned in the previous section. Both methods provide good tracking performance for the measurement points. It is also clear that the last three points have the largest displacement and hence also have the largest impact on the estimator results. The relative accuracy gets better as the displacement increases.

Finally the comparison of the estimated vertical force is shown in Fig. 8. This figure shows the very interesting result that even though both concept models provide good tracking of the measurements, the input forces for the trailing arm model are considerably underestimated. In order to clarify why this occurs, an additional estimator results is shown. In this estimator the modal reduced model is used without the application of the five other input loads shown in Fig. 5. This estimator provides very similar results to the trailing arm model, because considerable bias errors occur due to the neglected loads. These results clearly shows the importance of taking all the inputs into account accurately in a realistic structure like the twistbeam used here. Even though the use of simple models can be valid for some estimation problems, in the case of coupled loading, all these load effects should be taken into account and one degree-of-freedom models are insufficient. The modal reduced model which takes all load conditions into account leads to very accurate results and can be used to accurately infer the external forces applied to the system.

5 Conclusion

In this work the use of different concept models for a twistbeam suspension are discussed for input force estimation purposes. These techniques could be highly valuable in future active vehicle systems where more knowledge needs to be obtained about the state and environment of the vehicle. In practice it is very difficult to directly measure external loads on a system, especially in dynamic loading conditions. In order to meet this issue, this work proposes the use of concept models in an estimator for input forces. By coupling a model with an estimator, a Kalman filter in this case, the dynamic behavior of the structure can be taken into account and inference of dynamic force is possible as well. In order to estimate the forces concurrently with the states, an augmented approach is adopted in which the unknown inputs are added as additional states. For the model, two different concept models based on an updated finite element model are proposed. The first concept model considers one side of the twistbeam as a rigid trailing arm structure with only one input, being the vertical force. The second concept model is a modal reduced model where all external loads on the twistbeam are taken into account. Both models lead to a very compact model which can be exploited in online estimation problems. Experimental validation on the twistbeam system shows that good tracking of the measurements is achieved by both models, but the estimation of the input forces poses more important problems. It is demonstrated that it is necessary to take the full loading condition into account in order to have accurate estimates for the vertical forces. Because the trailing arm model is not capable of incorporating the other external loads, it leads to strongly biased estimates for the external forces. This case highlights the importance of proper model construction for estimation purposes. It is important to notice that this behavior can be predicted based on simulations with more accurate models and it is not necessary to perform measurements to find these effects. In the current work the other external loads, besides the vertical force, are collected through measurements and fed to the estimator. However, in practice this will not be possible and future research will focus on how all external loads can be estimated concurrently.

Acknowledgements

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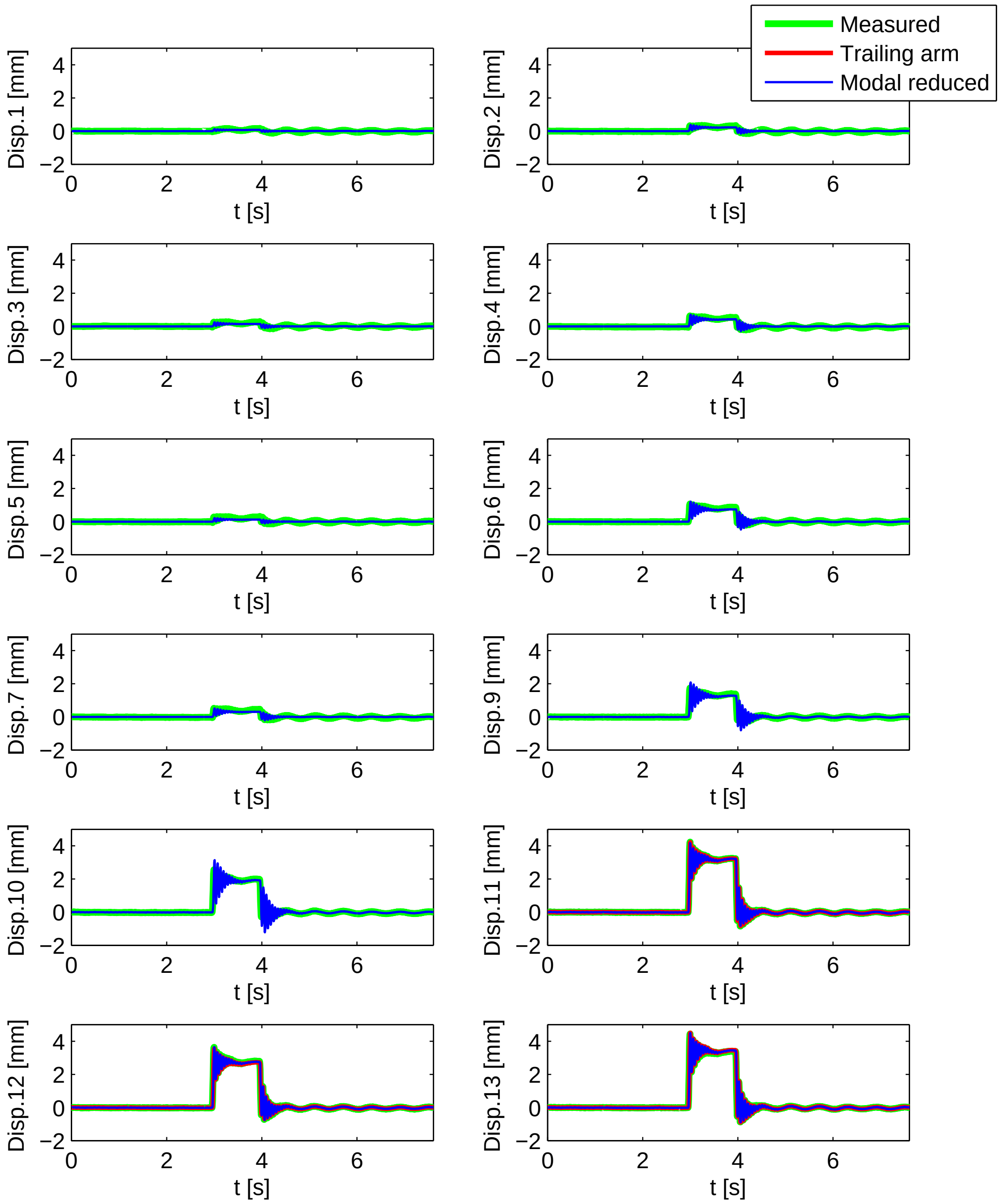

Figure 7: Tracking of the measurements for the estimator with concept models.

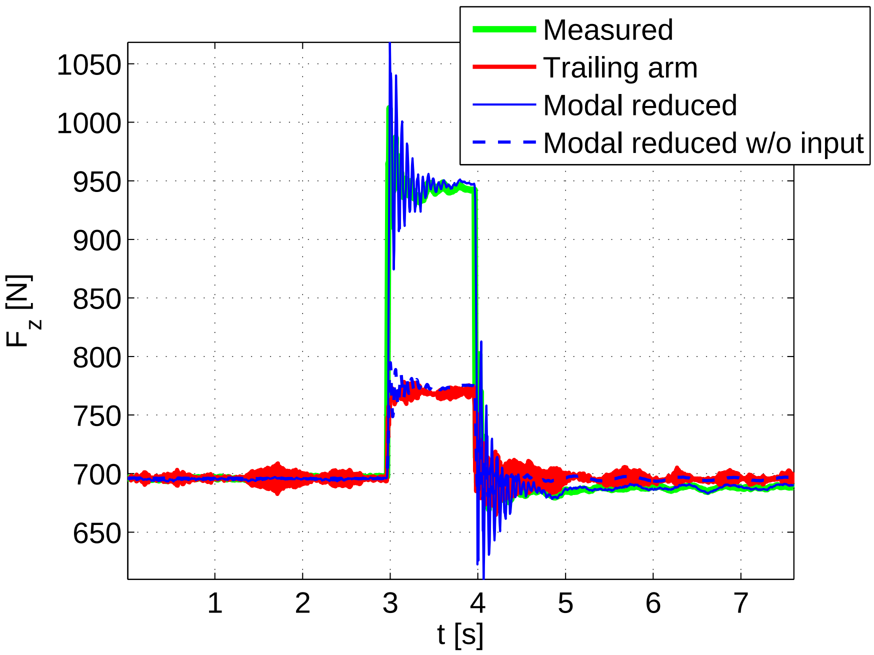

Figure 8: Comparison of estimated vertical forces.

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